SPECIFICATION

TITLE OF THE INVENTION

CONTINUOUSLY VARIABLE TRANSMISSION

BACKGROUND OF THE INVENTION

The present invention relates to a continuously variable transmission to be mounted on a vehicle and, specifically, to a technique effectively applied to a continuously variable transmission having a rubber drive belt.

A belt-driven continuously variable transmission (CVT) applied to a power transmission system of the vehicle has a primary pulley provided on an input shaft and a secondary pulley provided on an output shaft, wherein the drive belt is provided to extend for winding between the two pulleys. By changing the contact diameter of the drive belt, a rotation speed transmitted from the input shaft to the output shaft is changed continuously.

In such a continuously variable transmission, the contact diameter of the drive belt is changed. Therefore, there are a hydraulic type one in which a groove width of the primary pulley is variably controlled by hydraulic pressure in accordance with a run condition, and a weight type one in which a groove width of the primary pulley is variably controlled by centrifugal weight in accordance with the rotation speed of the primary pulley.

The hydraulic-type continuously variable transmission is more suitable than the weight-type one to high-precisely control a transmission gear ratio depending on the run condition. However, the hydraulic-type continuously variable transmission requires a

hydraulic control device, which performs hydraulic control in accordance with the run condition of the vehicle, so that adoption of the hydraulic-type continuously variable transmission involves cost increase. In contrast, the weight-type continuously variable transmission is capable of being changed in accordance with the rotation speed of the primary pulley, so that its construction is simple and, consequently, the manufacturing costs of the continuously variable transmission can be reduced. For these reasons, the weight-type continuously variable transmission is mounted on all terrain vehicles (ATVs) and two-wheel vehicles, etc. in many cases.

A rubber drive belt is built in the weight-type continuously variable transmission, and the drive belt requires being cooled for preventing its deterioration and improving its durability. Therefore, there has been developed a continuously variable transmission in which, by providing fan blades on the primary pulley, cooling air is blown into a case in which the pulley and drive belt are accommodated (see, for example, Japanese Patent Laid-open (TOKUKAIHEI) 11-11171).

SUMMARY OF THE INVENTION

However, the diameter sizes of the primary pulley and secondary pulley on which the fan blades are formed are limited, and the length of each fan blade has been difficult to increase. Particularly, to increase the length of each fan blade beyond the diameter size of each pulley and/or to increase the width size of each fan blade lead to the unnecessarily jumbo-sized continuously variable transmission.

Since the jumbo-sizing of the fan blades to be formed on the pulleys is thus restricted, it is difficult to increase a flow rate of the cooling air for cooling a interior of the case. Therefore, the drive belt cannot be sufficiently cooled, whereby there is the problem that the durability of the drive belt is degraded.

An object of the present invention is to improve the durability of a continuously variable transmission by sufficiently cooling the interior of the case of the continuously variable transmission.

A continuously variable transmission continuously according to the present invention, which changes rotation of a primary pulley driven by an engine and transmits the rotation to a secondary pulley through a drive belt, comprises: a fan blade sending cooling air to said pulleys and said belt and provided to at least one of said primary pulley and said secondary pulley; and a scroll surface formed in a case rotatably accommodating said primary pulley and said secondary pulley from an intake region of the cooling air toward a discharge region thereof so as to gradually away from a top face of said fan blade in a radial-outer direction.

A continuously variable transmission according to the present invention claim further comprises: an intake port for introducing the cooling air into said case; and an exhaust port for exhausting the cooling air therefrom, wherein the intake and exhaust ports are formed in said case.

A continuously variable transmission according to the present invention further comprises: an unidirectional airflow plate

provided in said case and making unidirectional the cooling air introduced onto said scroll surface to a rotational direction of said fan blade.

According to the present invention, the scroll surface is formed from the cooling-air intake region to the cooling-air discharge region so as to be gradually away from the top faces of the fan blades. Consequently, back pressure disturbing the flow of the discharged cooling air can be suppressed, whereby the blowing efficiency of the cooling air can be enhanced.

Therefore, since the interior of the case can be sufficiently cooled, the durability of the drive belt can be enhanced and the durability of continuously variable transmission can be also enhanced.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a perspective view showing a vehicle.

FIG. 2 is a schematic view showing an engine unit and a drive unit that are mounted in the vehicle.

FIG. 3 is a cross-sectional view taken along line A-A of FIG. 2.

FIG. 4 is a partially enlarged cross-sectional view showing a continuously variable transmission shown in FIG. 2.

FIG. 5 is a side view showing the continuously variable transmission as viewed from arrow A of FIG. 4.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Hereinafter, an embodiment of the present invention will be detailed with reference to the drawings. FIG. 1 is a perspective

view showing a vehicle, and this vehicle is an ATV generally called a "buggy", namely, an unleveled-ground traveling vehicle. As shown in FIG. 1, a vehicle body 1 is provided with front wheels 2a and 2b and rear wheels 3a and 3b, and a saddle-type seat 4 is provided in a central portion of the vehicle body 1. A rider rides on the vehicle by straddling the seat 4 and operates a steering handle 5 to make the vehicle run.

FIG. 2 is a schematic view showing an engine unit 10 and a drive unit 11 that are mounted in the vehicle in FIG 1. FIG. 3 is a cross-sectional view taken along line A-A of FIG. 2. As shown in FIGs. 2 and 3, the engine unit 10 for outputting an engine power is provided on a vehicle-front side. On a vehicle-rear side, there is provided the drive unit 11 transmitting the engine power to drive wheels via a continuously variable transmission 55 according to an embodiment of the present invention.

As shown in FIG. 2, a crankshaft 13 is rotatably accommodated via a bearing in a crankcase 12 of the engine unit 10. Additionally, as shown in FIG. 3, a cylinder 14 is attached to an opening formed on the crankcase 12, and a cylinder head 15 is mounted on an upper end face of the cylinder 14. A piston 16 is reciprocatably built in a cylinder bore formed in the cylinder 14. A piston pin 17 attached to the piston 16 and a crank pin 18 fixed to the crankshaft 13 to be eccentric to a rotational center of the crankshaft 13 are linked to each other via a connecting rod 19.

In the cylinder head 15, a combustion chamber 15a is formed, and an intake port 15b and an exhaust port 15c are formed to open to the combustion chamber 15a. An intake valve 20 is built in the cylinder head 15 so that the intake port 15b and the combustion

chamber 15a can be shifted from communication conditions to cutoff conditions or vice versa. An exhaust valve 21 is built in the cylinder head 15 so that the intake port 15b and the combustion chamber 15a can be shifted from communication conditions to cutoff conditions or vice versa.

Additionally, a camshaft 22 having two cam surfaces is rotatably mounted in the cylinder head 15. On a locker shaft 23 provided in parallel thereto, a locker arm 23a for open/close-driving the intake valve 20 and a locker arm 23b for open/close-driving the exhaust valve 21 are rotatably mounted. An unshown timing chain is provided to extend for winding between an unshown sprocket fixed to the camshaft 22 and a sprocket 24 shown in FIG. 2 and fixed to an end of the crankshaft 13, whereby the camshaft 22 is rotate-driven in synchronization with the rotation of the crankshaft 13. The cam surface of the camshaft 22 is made to contact with one ends of the locker arms 23a and 23b in accordance with rotation positions of the crankshaft 13, that is, displacement positions of the piston 16. Accordingly, each of the intake valve 20 and the exhaust valve 21 is open/close-driven at a predetermined timing.

An engine 25 including the above-mentioned crankcase 12, cylinder 14, and cylinder head 15 is mounted on the vehicle body 1 so that the crankshaft 13 is faced in a vehicle-width direction. The engine 25 is a single-cylinder air-cooled engine, and heat-releasing fins 26 are formed on the cylinder 14 and the cylinder head 15.

A carburetor 27 is disposed on the vehicle-rear side of the engine 25 since the engine 25 is driven by supplying an air-fuel

mixture to the engine 25. An input port 28a of the carburetor 27 is connected to an unshown air cleaner, and an output port 28b of the carburetor 27 is connected via an intake pipe 32 to the intake port 15b of the cylinder head 15. One end of a throttle cable 30 is assembled to the carburetor 27, and the other end of the throttle cable 30 is assembled to an acceleration grip 6 shown in FIG. 1. Additionally, a fuel hose 31 for guiding fuel from a fuel tank 7 shown in FIG. 1 is connected to the carburetor 27.

The air-fuel mixture guided from the carburetor 27 to the intake port 15b in accordance with a rider's operation of the acceleration grip 6 is fed into the combustion chamber 15a in an intake stroke in which the intake valve 20 is open-driven, and is converted to the engine power by burning through a compression stroke and a combustion stroke. The burned air-fuel mixture becomes an exhaust gas and is exhausted, in an exhaust stroke, from the exhaust port 15c to the outside through an unshown exhaust pipe. The piston 16 pushed down due to the burning of the air-fuel mixture rotate-drives the crankshaft 13 via the connecting rod 19, whereby the engine power is output to the drive unit 11 described below.

As shown in FIG. 3, two balancer shafts 40 and 41 are rotatably attached to the crankcase 12 via bearings. Balancer weights 40a and 41a are integrally provided on the balancer shafts 40 and 41, respectively. Gears 40b and 41b provided on the respective balancer shafts 40 and 41 engage with a gear 42 provided on the crankshaft 13, whereby a rotational deviation of the crankshaft 13 is absorbed by the balancer weights 40a and 41a. Note that, in FIG. 2, the balancer shaft 40 that is one of the two

balancer shafts is shown.

An oil pump 43 driven by the crankshaft 13 is provided to the one end of the crankshaft 13, and lubricating oil discharged from the oil pump 43 is supplied to a sliding portion of the drive unit 11 through an unshown oil path. Further, an electric generator 44 driven by the crankshaft 13 is provided to the other end of the crankshaft 13, and electric power generated by the electric generator 44 charges an unshown battery. Additionally, a starter motor 45 is provided to be adjacent to the electric generator 44, whereby the rotation of the starter motor 45 driven at an engine start is transmitted to the crankshaft 13 via gears 46a and 46b.

As shown in FIG. 2, in the crankcase 12, a countershaft 47 is rotatably mounted in parallel to the crankshaft 13. A gear 48a provided on the countershaft 47 engages with a gear 48b provided on the crankshaft 13, whereby the rotation of the crankshaft 13 is transmitted to the countershaft 47. A recoil cover 49 is built in the crankcase 12 disposed on one end side of the countershaft 47. In the recoil cover 49, there is mounted a recoil starter 50 used for starting manually the engine 25 on the case where the engine 25 is difficult to start up due to lack of the power amount charged in the buttery. The recoil starter 50 has: a recoil pulley 50b, which is accommodated in the recoil cover 49 and around which a recoil rope 50a is wound; and a recoil drum 50c attached to the countershaft 47. Since the recoil pulley 50b is rotated by pulling the recoil rope 50a, the crankshaft 13 is rotated via the countershaft 47 and therefore the engine 25 can be started.

Also, a centrifugal clutch 51 is attached to the other end of the countershaft 47. The centrifugal clutch 51 has a clutch drum 51a rotatably mounted in the crankcase 12 and a rotary plate 51b fixed to the countershaft 47. A plurality of arcuate clutch shoes 51c are mounted on the rotary plate 51b. Each of the clutch shoes 51c is rotatable via a pin 51d attached to one end thereof. A tensile coil spring 51e is attached to the other end of the clutch shoe 51c, whereby a spring force is exerted on the clutch shoe 51c in such a direction as to be away from an inner circumferential surface of the clutch drum 51a. Accordingly, when the rotation speed of the countershaft 47 exceeds a predetermined one, a centrifugal force applied to the clutch shoe 51c exceeds the spring force, whereby the clutch shoe 51c engages with the inner circumferential surface of the clutch drum 51a and the centrifugal clutch 51 becomes in a fastening condition. Consequently, the engine power from the crankshaft 13 is transmitted to the clutch drum 51a via the countershaft 47, and the engine power is input from the clutch drum 51a into the continuously variable transmission 55.

The continuously variable transmission 55 has a transmission case 53 assembled in the crankcase 12, and the transmission case 53 comprises a case body 53a and a case cover 53b. A primary shaft 52 fixed to the clutch drum 51a and a secondary shaft 54 placed in parallel to the primary shaft 52 are rotatably accommodated in the transmission case 53.

Further, the continuously variable transmission 55 includes a primary pulley 56 provided on the primary shaft 52 and a secondary pulley 57 provided on the secondary shaft 54. The

primary pulley 56 has a conical-surface-shaped fixing sheave 56a and a conical-surface-shaped moving sheave 56b opposite to the fixing sheave 56a. The fixing sheave 56a is fixed to the primary shaft 52, and the moving sheave 56b is movably attached axially to a spline provided on the primary shaft 52. Meanwhile, the secondary pulley 57 has a conical-surface-shaped fixing sheave 57a and a conical-surface-shaped moving sheave 57b opposite to the fixing sheave 57a. The fixing sheave 57a is fixed to the secondary shaft 54. The moving sheave 57b is movably attached axially to a spline provided on the secondary shaft 54.

A V-belt 60 serving as a rubber drive belt is provided to extend for winding between the primary pulley 56 and the secondary pulley 57. When a contact diameter of the V-belt 60 with regard to the primary pulley 56 and the secondary pulley 57 is changed, the rotation speed of the primary shaft 52 is continuously changed and transmitted to the secondary shaft 54. On the moving sheave 56b of the primary pulley 56, a plurality of columnar centrifugal weights 61, for example, six column centrifugal weights 61 facing perpendicularly to the rotational center of the primary shaft 52 are mounted. A cam surface 62 corresponding to each surface of the centrifugal weights 61 is formed on the moving sheave 56b. The cam surface 62 is formed so that a radial-outer side portion of the moving sheave 56b protrudes toward an end of the primary shaft 52. A cam plate 63 is fixed to the primary shaft 52 so as to be opposite to the cam surface 62, and a radial-outer side portion of the cam plate 63 inclines so as to approach to the cam surface 62. Meanwhile, a spring seat 64 is fixed to the secondary shaft 54. A compression coil spring 65 for applying a fastening

force to the V-belt 60 is mounted between the spring seat 64 and the moving sheave 57b.

As the rotation speed of the primary shaft 52 is increased, a centrifugal force exerted on each of the centrifugal weights 61 is increased in strength. Therefore, each centrifugal weight 61 moves in a radial-outer direction while pushing and extending a space between the moving sheave 56b and the cam plate 63. In this case, since the cam plate 63 is fixed to the primary shaft 52, the moving sheave 56b approaches toward the fixing sheave 56a due to the movement of the centrifugal weights 61. For this reason, since the groove width of the primary pulley 56 is narrowed, the contact diameter of the V-belt 60 with regard to the primary pulley 56 becomes large. Meanwhile, since the groove width of the secondary pulley 57 is enlarged against the spring force due to the V-belt 60, the contact diameter of the V-belt 60 with regard to the secondary pulley 57 becomes small. In other words, as the rotation speed of the primary shaft 52 increases, the speed ratio of the continuously variable transmission 55 is shifted to a high-speed side.

Further, when the rotation speed of the primary shaft 52 is reduced and the centrifugal force to be exerted on each centrifugal weight 61 becomes small in strength, the groove width of the secondary pulley 57 is narrowed by the spring force exerted on the secondary pulley 57. Therefore, the contact diameter of the V-belt 60 with regard to the secondary pulley 57 becomes large. Meanwhile, since the groove width of the primary pulley 56 is enlarged due to the V-belt 60, the contact diameter of the V-belt 60 with regard to the primary pulley 56 becomes small. That is,

as the rotation speed of the primary shaft 52 reduces, the speed ratio of the continuously variable transmission 55 is shifted to a low-speed side.

One end of the secondary shaft 54 protrudes from the transmission case 53, and is supported via bearings in a gear case 66 built in the transmission case 53. In the gear case 66, an output shaft 67 is rotatably accommodated parallel to the secondary shaft 54 and, simultaneously, a wheel shaft 68 is rotatably mounted parallel to the output shaft 67.

A forward gear 69a is integrally provided on the secondary shaft 54, wherein the forward gear 69a engages always with a gear 69b rotatably mounted on the output shaft 67. Additionally, a rearward sprocket 70a is integrally provided on the secondary shaft 54, wherein a chain 70c is provided to extend for winding between the rearward sprocket 70a and a sprocket 70b rotatably mounted on the output shaft 67. Accordingly, a rotational direction of the gear 69b gear-driven by the power supplied from the secondary shaft 54 is opposite to that of the secondary shaft 54, and the rotational direction of the chain-driven sprocket 70b is identical with that of the secondary shaft 54.

Further, a forward/rearward shift mechanism 71 is mounted between the gear 69b and the sprocket 70b. In response to a shift operation of the forward/rearward shift mechanism 71, the powers supplied from the gear 69b and the sprocket 70b are selectively transmitted to the output shaft 67. The forward/rearward shift mechanism 71 has a pair of shift discs 72a and 72b that engage with the respective splines of the output shaft 67. The shift discs 72a and 72b are axially slidable on the output shaft 67.

Engagement gear teeth 73b, which engage with engagement gear teeth 73a provided on a side face of the gear 69b, are provided on the shift disc 72a, and engagement gear teeth 74b, which engage with engagement gear teeth 74a provided on a side face of the sprocket 70b, are provided on the shift disc 72b. Accordingly, when the pair of shift discs 72a and 72b are moved toward the gear 69b and the engagement gear teeth 73a and 73b are engaged with one another, the rotation of the secondary shaft 54 is transmitted to the output shaft 67 via the forward gears 69a and 69b. Meanwhile, when the shift discs 72a and 72b are moved toward the sprocket 70b and the engagement gear teeth 74a and 74b are engaged with one another, the rotation of the secondary shaft 54 is transmitted to the output shaft 67 via the rearward sprockets 70a and 70b. Note, as shown in FIG. 2, if each of the shift discs 72a and 72b is not engaged with any of the engagement gear teeth, linkage between the secondary shaft 54 and the output shaft 67 is cut off.

Additionally, a pair of shift discs 75a and 75b engaging with the respective splines of the output shaft 67 are slidably attached axially to the output shaft 67. Engagement gear teeth 76b, which engage with engagement gear teeth 76a provided on the gear case 66, are proved on the shift disc 75b. Accordingly, when the shift discs 75a and 75b are moved toward the gear case 66 and the engagement gear teeth 76a and 76b are engaged with one another, the output shaft 67 and the gear case 66 are fastened, whereby the rotation of the output shaft 67 is restricted. Meanwhile, as shown in FIG. 2, when the engaging of the engagement gear teeth 76a and 76b is released, the output shaft 67 becomes in a rotatable state.

The shifting of such shift discs 72a, 72b, 75a, and 75b is carried out by shift holders 77 and 78. The shift holders 77 and 78 are linked via an unshown operation link to a shift lever 8 shown in FIG. 1. The shift discs 72a, 72b, 75a, and 75b are shifted by the operation of the shift lever 8 by the rider. In the shift lever 8, there are set: an F position corresponding to a forward run; an R position corresponding to a rearward run; an N position corresponding to a neutral mode of the drive unit 11; and a P position corresponding to a parking mode of the vehicle.

If the shift lever 8 is operated and shifted to the F position, the engagement gear teeth 73b of the shift disc 72a are engaged with the engagement gear teeth 73a of the gear 69b and the shift discs 75a and 75b are each shifted to a neutral position. Further, when the shift lever is operated and shifted to the R position, the engagement gear teeth 74b of the shift disc 72b are engaged with the engagement gear teeth 74a of the sprocket 70b and the shift discs 75a and 75b each become at a neutral position. Further, when the shift lever is operated and shifted to the N position, all the shift discs 72a, 72b, 75a, and 75b become at the neutral positions. When the shift lever is operated and shifted to the P position, the shift discs 72a and 72b each become at the neutral position and the engagement gear teeth 76b of the shift disc 75b are engaged with the engagement gear teeth 76a of the gear case 66.

A gear 79a is fixed to the output shaft 67 to which the power is transmitted in accordance with the operation of the above-mentioned shift lever 8, and a gear 79b always engaging with the gear 79a is fixed to wheel shaft 68. Rear wheels 3a and 3b

are linked to both ends of the wheel shaft 68, and the rear wheels 3a and 3b serving as drive wheels are driven by the wheel shaft 68. Note that, as shown in FIG. 3, a drive shaft 80 driving the front wheels 2a and 2b and provided with a gear 80a engaging with the gear 79b is rotatably supported by the transmission case 53 and the gear case 66 and that a front-wheel output shaft 81 linked to the drive shaft 80 via a bevel gear 81a is rotatably supported by the gear case 66. Thus, the power from the output shaft 67 is transmitted to the front-wheel output shaft 81 via the drive shaft 80, so that the front wheels 2a and 2b are driven together with the rear wheels 3a and 3b.

Additionally, to brake the vehicle at the time of a run, a brake disc 82 is mounted on the output shaft 67, as shown in FIG. 2. To the gear case 66, there is attached a brake caliper 83 engaging a brake pad 83a with the brake disc 82. The brake caliper 83 is driven in the manner that the rider operates a brake lever 9 provided to the steering handle 5, whereby a braking force can be applied to the output shaft 67.

Next, a cooling structure for the continuously variable transmission 55 will be described. FIG. 4 is a partially enlarged cross-sectional view showing the continuously variable transmission 55 shown in FIG. 2. FIG. 5 is a side view showing the continuously variable transmission 55 as viewed from arrow A of FIG. 4 and illustrates the condition where the case cover 53b is released. Note that outline arrows on a colored background, as illustrated in FIGs. 4 and 5, indicate flow directions of cooling the air flowing in the transmission case 53, and arrows "A" illustrated in FIG. 5 indicate respective rotational directions of

the pulleys 56 and 57.

As shown in FIGs. 4 and 5, outside air is introduced as the cooling air into the transmission case 53 in order to cool the primary pulley 56, the secondary pulley 57, and the V-belt 60 that constitute the continuously variable transmission 55. Into the case body 53a of the transmission case 53, an intake port 85a is formed to introduce the cooling air. The intake port 85a communicates with the outside, from a flow passage in the crankcase 12, via an intake duct 85b. Further, an exhaust port 86a for exhausting the cooling air is formed in the case cover 53b of the transmission case 53. The exhaust port 86a communicates with the outside, via an exhaust duct 86b formed in the case cover 53b. Note that unshown extension ducts are connected to the intake duct 85b and the exhaust duct 86b, whereby intrusion of water, dust, and the like into the transmission case 53 can be prevented.

To suck the cooling air from the intake port 85a into the transmission case 53 and to exhaust the cooling air having absorbed heat from the V-belt 60 and the like, a plurality of fan blades 87 are formed on a rear surface of the fixing sheave 56a of the primary pulley 56 so that they extend in a radial-outer direction. Additionally, fan blades 88a and 88b are formed on rear surfaces of the fixing sheave 57a and the moving sheave 57b of the secondary pulley 57 so that they extend in the radial-outer direction.

As shown in FIG. 5, a shroud wall 90 having a scroll surface 89 along top faces 87a of the fan blades 87 is formed in the case body 53a and near the intake port 85a formed in the case body 53a

of the transmission case 53, more specifically, within an intake region Ai where the fan blades 87 suck the cooling air. The scroll surface 89 is smoothly and continually formed on an inner circumferential surface of the case body 53a from the shroud wall 90 toward a discharge region Ao where the fan blades 87 discharge the cooling air. More specifically, the scroll surface 89 is formed from the intake region Ai toward the discharge region Ao to be gradually away from the top faces 87a of the fan blades 87 in the radial-outer direction. That is, a clearance C1 is formed between the top faces 87a of the fan blades 87 and the scroll surface 89 in the intake region Ai while a clearance C2 larger than the clearance C1 is formed in the discharge region Ao.

Also, a unidirectional-airflow plate 91 for making unidirectional the cooling air guided onto the scroll surface 89 is formed in the discharge region Ao of the case body 53a. The unidirectional-airflow plate 91 is attached in such a manner as to partition a flow path of the cooling air flowing between the top faces 87a of the fan blades 87 and the scroll surface 89, whereby the cooling air can be prevented from diffusing unnecessarily by making unidirectional the cooling air in the rotational direction of the fan blades 87.

Subsequently, a cooling process using the cooling air flowing in the transmission case 53 during the driving of the continuously variable transmission 55 will be described. After the engine 25 is started by the operation of the starter motor 45, the vehicle is shifted to a forward-run condition by the rider's operation of shifting the shift lever 8 to the F position. Under such a condition, since the acceleration grip 6 is operated, the

centrifugal clutch 51 is shifted to the fastening condition as the engine speed is raised and the vehicle starts a forward run.

As shown in FIG. 5, in the run condition of the vehicle, the primary pulley 56 is rotate-driven in the direction of the arrow A and the secondary pulley 57 is also rotate-driven in the direction of the arrow A. Concurrently, the cooling air is sent from the intake port 85a of the transmission case 53 to the exhaust port 86a by the fan blades 87, 88a, and 88b formed on the respective pulleys 56 and 57. Note that even in the rear-run condition, since the pulleys 56 and 57 rotate in the direction of the arrow A, the cooling air is similarly sent also in the rear-run condition.

First, the cooling air collected by the fan blades 87 in the intake region Ai of the primary pulley 56 is pressurized and sent to the discharge region Ao by the rotation drive of the primary pulley 56 and, then, sent from the discharge region Ao to the secondary pulley 57. At this time, the fan blades 87 can securely collect the cooling air having passed through the intake port 85a by using the scroll surface 89 of the intake region Ai formed along the top faces 87a of the fan blades 87. Additionally, by the scroll surface 89 formed from the intake region Ai toward the discharge region Ao so as to be gradually away from the top faces 87a of the fan blades 87, back pressure disturbing the flow of the cooling air discharged from the fan blades 87 can be suppressed, whereby blowing efficiency of the cooling air can be enhanced. Further, since the unidirectional-airflow plate 91 prevents the cooling air from unnecessarily diffusing, the cooling air can be securely supplied to the secondary pulley 57.

Next, as shown in FIG. 4, the cooling air discharged from

the fan blades 87 of the primary pulley 56 flows to the secondary pulley 57 while spreading in the transmission case 53 in the width direction. After cooling the primary pulley 56, the secondary pulley 57, and the V-belt 60 that have generated heat due to friction generated at the time of transmitting the power, the cooling air is exhausted to the outside through the exhaust port 86a. Even when the cooling air is exhausted through the exhaust port 86a, the cooling air is sent toward the exhaust port 86a by the fan blades 88a and 88b formed on the secondary pulley 57. Consequently, since a pressure rise around the secondary pulley 57 can be suppressed, the blowing efficiency of the cooling air can be further enhanced.

As described above, since the blowing efficiency of the cooling air flowing in the transmission case 53 is enhanced, the primary pulley 56, the secondary pulley 57, and the V-belt 60 can be sufficiently cooled and the durability of the continuously variable transmission 55 can be improved. Particularly, in the continuously variable transmission 55 having the rubber V-belt 60 whose deterioration is accelerated by heat, the durability of the V-belt 60 can be enhanced and the running costs of the continuously variable transmission 55 can be reduced. Additionally, the blowing efficiency of the cooling air can be enhanced without jumbo-sizing the fan blades 87, 88a, and 88b. Therefore, the jumbo-sizing and/or the cost increase of the continuously variable transmission 55 can be suppressed.

The present invention is not limited to the above-mentioned embodiment, and can be variously modified and altered without departing from the gist thereof. For example, although the

continuously variable transmission 55 is mounted on the ATV, that is, the unleveled-ground traveling vehicle, the continuously variable transmission 55 may be mounted on a two-wheeled vehicle and the like.

Further, although the scroll surface 89 formed in the transmission case 53 is formed on a side of the primary pulley 56, the scroll surface 89 may be formed on a side of the secondary pulley 57. In this case, the scroll surface is formed in the vicinity of the exhaust port 86a which is within the discharge region of the secondary pulley 57 so as to be gradually away from the top faces of the fan blades 88a and 88b.

Additionally, in the above-mentioned embodiment, the cooling air is sent to the secondary pulley 57 from the primary pulley 56. However, the cooling air may be sent from the secondary pulley 57 to the primary pulley 56 by forming an intake port on the side of the secondary pulley 57 and forming an exhaust port on the side of the primary pulley 56.

Further, in the drawings, the respective fan blades 87, 88a, and 88b formed on the primary pulley 56 and the secondary pulley 57 are illustrated as linearly extending radial fans. However, such fan blades are not limited to the above-mentioned embodiment, and may be centrifugal fans arcuately formed to have convex surfaces in a rotational direction or may be multi-blade fans arcuately formed to have convex surfaces in the rotational direction.

Note that, in the above-mentioned embodiment, the fan blades 87, 88a, and 88b are formed on the primary pulley 56 and the secondary pulley 57, respectively. However, needless to say, the

fan blades 87 may be formed only on the primary pulley 56, or the fan blades 88a and 88b may be formed only on the secondary pulley 57.

According to the present invention, the scroll surface is formed from the cooling-air intake region to the cooling-air discharge region so as to be gradually away from the top faces of the fan blades. Consequently, back pressure disturbing the flow of the discharged cooling air can be suppressed, whereby the blowing efficiency of the cooling air can be enhanced.

Therefore, since the interior of the case can be sufficiently cooled, the durability of the drive belt can be enhanced and the durability of continuously variable transmission can be also enhanced.

The entire disclosure of a Japanese Patent Application No. 2003-114733, filed on April 18, 2003 including specification, claims, drawings and summary, on which the Convention priority of the present application is based, are incorporated herein by reference in its entirety.